



Thermo-economic analysis of a micro-cogeneration system based on a rotary steam engine (RSE)

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ABSTRACT

A rotary steam engine (RSE) is a simple, small, quiet and lubricant-free option for micro-cogeneration. It is capable of exploiting versatile thermal sources and steam temperatures of 150–180 °C, which allow operational pressures less than 10 bar for electrical power ranges of 1–20 kW_e. An RSE can be easily integrated in commercially available biomass-fired household boilers. In this paper, we characterize the boiler-integrated RSE micro-cogeneration system and specify a two-control-volume thermodynamic model to conduct performance analyses in residential applications. Our computational analysis suggests that an RSE integrated with a 17 kW_{th} pellet-fueled boiler can obtain an electrical output of 1.925 kW_e in the design temperature of 150 °C, the electrical efficiency being 9% (based on the lower heating value of the fuel, LHV) and the thermal efficiency 77% (LHV). In a single-family house in Finland, the above system would operate up to 1274 h/y, meeting 31% of the house's electrical demand. The amount of electricity delivered into the grid is 989 kW h/y. An economic analysis suggests that incremental costs not exceeding € 1500 are justifiable at payback periods less than five years, when compared to standard boilers.

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1. Introduction

The most cited micro-cogeneration technologies within the power range of 1–30 kW_e are micro-turbines (MT), fuel cells (FC), Stirling engines (SE), and internal combustion engines (ICE) [1]. Within higher power ranges, such as 100 kW_e to 1 MW_e, steam cycles and combined cycles have been used (e.g. Badami et al. [2]). Micro-turbines in the smallest scale (less than 30 kW_e) suffer from low electrical efficiencies, at around 6–8% [3]. Both micro-turbines and fuel cells require pre-processed refined fuels (e.g. through electrolysis or gasification), which easily results in costly extra investments particularly, when biomass is utilized as a fuel [4–7]. Stirling engines (SE) fill a niche with their ability to employ a variety of thermal sources. Reductions up to 20% in primary energy and emissions [8] and 28% self-sufficiency in the daily electrical demand [9] for residential applications have been reported. Micro-cogeneration based on a Stirling engine in the smallest scale (1 kW_e) is plagued by relatively low electrical efficiency (~10%) and high thermal to electrical power ratio, which limits its applicability in buildings in the future, their electrical demands being significant with respect to thermal demands. The internal combustion engine (ICE) is a time-tested technology with an

established market [10]. About 30,000 micro-CHP units based on 5 kW_e ICE's have been installed in Germany in the recent years [11]. Average thermal efficiencies of 76% and electrical efficiencies of around 12% have been achieved. Aussant et al. [12] and Ren & Gao [13] have concluded that greenhouse gas emissions can be reduced in residential applications by using ICE technology, although an upper limit in reductions is quickly realized since fossil fuels are predominant.

Requirements related to thermal sources, efficiency, operational temperatures, controllability, noise, space requirement and capital cost call for the introduction of new micro-cogeneration technologies. Bonnet et al. [14] suggest the Ericsson engine as an alternative to SE due to its less complicated design and thus potentially reduced costs. Organic Rankine Cycle (ORC) has been considered a promising alternative for micro-cogeneration and especially suitable for biomass [15,16]. Low temperature requirements have been pointed out as the major benefit of ORC [17,18], and hybridization with a solar desalination system has been proposed [19]. The investment costs of an ORC system have been estimated to be up to 60% less than those of SE [20]. Quoilin et al. [21] conclude that the specific investment cost of an optimized micro-scale (<5 kW_e) ORC system (including pump, evaporator, condenser, expander, working fluid and other components) lies above € 2000/kW_e. The investment cost is sensitive to both the evaporator temperature and the working fluid. Qiu et al. [22] note that there are not yet

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commercially available expanders for micro-scale ORC systems, but suggest the use of scroll and vane expanders in these applications.

The rotary steam engine (RSE) is a long-established concept based on the invention by James Watt in the late 1700s [23]. An RSE offers the capability of applying versatile thermal sources and low operational temperatures and pressures with a noise level and minimal service requirements. Since 2006, the Aalto University School of Engineering has been developing a new rotary steam engine based on patented innovations that utilize low temperature levels (150–180 °C). The rotary steam engine is an oil-free medium-speed (1000–2000 RPM) engine, which can utilize either solar heat or combustion gases as a heat source for the steam cycle. The main application will be a 1–10 kW_e micro-cogeneration plant, especially employed to utilize biomass as a heat source.

This paper outlines a simple, novel micro-cogeneration system, where an RSE is integrated with a conventional domestic wooden-pellet-fired boiler. Using a two-control-volume thermodynamic model, we match the proper evaporator and condenser temperatures and the compression ratio with the maximum electrical efficiency of the RSE. Moreover, we conduct a computational analysis to characterize the operational performance of a 17 kW_{th} RSE boiler. We also determine the daily operational hours and energy balance of the RSE boiler in a typical single-family house in Finland. We conclude by discussing the future development and research needs for the commercialization of the system.

2. Integrated RSE micro-cogeneration system

2.1. RSE process

Thermodynamically, the RSE process is a Rankine cycle, where saturated pressurized vapor expands in the engine and is then condensed in vacuum conditions in a condenser.

The fundamental difference compared to the ordinary Rankine cycle is that the final pressure p_2 in the RSE after the expansion is not necessarily the condenser pressure. The pressure p_2 depends on expansion volume ratio V_2/V_1 , which is a function of volume V_1 (filled with constant pressure steam) and of final volume V_2 , which is the final volume of the vapor after the expansion within the expanding closed chamber (see Fig. 1). After the expansion, the outlet valve opens, and the steam flows from the RSE to the condenser, achieving the final condenser pressure p_3 .

In Fig. 1, the white area above p_{am} corresponds to the work done by the engine when the condenser is working at ambient pressure conditions. The gray area (below p_{am}) is the additional work available when the condenser is working in vacuum conditions.

The advantages of an RSE are its compact size, a good power to weight ratio, and a simple connection to a rotational generator without gearing. An RSE does not need superheating of vapor; the construction of RSE's is more or less insensitive to liquid droplets that may form upon expansion. The required heat exchangers in an RSE are usually smaller than in a Stirling engine, because the heat transfer coefficients for boiling and condensation are larger than the convective heat transfer coefficient of SE working fluids such as helium. In addition, the outlet temperature of hot combustion gases after the boiler section is easily lower than in Stirling engines with the same size of a heat exchanger, which corresponds to a better utilization of primary heat. Moreover, the waste heat of an RSE is delivered as steam, which can be easily and effectively utilized e.g. in multi-stage flash (MSF) distillation or multi-effect distillation (MED) distillation plants.

2.2. RSE boiler system

The RSE is integrated into a hydronic heating system that circulates water through a pellet-fueled boiler. The conventional boiler

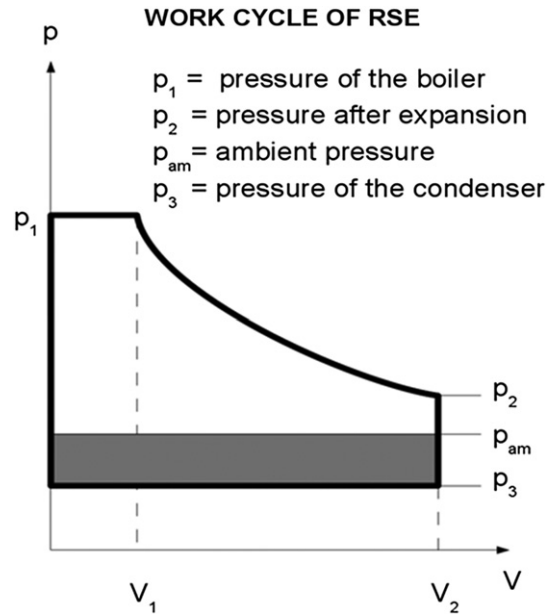


Fig. 1. PV-diagram of the work cycle of the RSE.

entails a furnace (combustion space) and a hot water tank, the temperature of which is kept between lower and upper boundaries (80–90 °C) by the ON–OFF control (TC1) of the heat source. The temperature of the supply water for the radiator network is controlled by mixing the return water from the radiator network with the hot boiler water, and the set point for the supply water temperature (max. 70 °C) is determined on the basis of outdoor temperature (TC2). Domestic hot water (55 °C) is also produced in the boiler, by circulating cold water through a heat exchanger located in the water tank of the boiler. Fig. 2 shows a schematic diagram [24].

For an integrated boiler, the furnace is replaced by the evaporator of the RSE. The return water from the radiator network is circulated through the condenser into the water tank of the boiler. The condenser can be located before or after the water tank, or inside the tank. The burner is controlled as above. When the burner is fired up, the RSE process is pressurized and the electricity generation starts.

The heat generation closely follows the thermal demand of the building. A pump circulates water from the condenser to the evaporator, taking its power from the shaft of the RSE. Any excess electricity is fed into a battery or the power grid through a power conditioning system at the required AC/DC voltage. A schematic diagram of the integrated boiler is shown in Fig. 3, and Fig. 4 presents a photo of an RSE prototype unit with power conditioning module.

To increase the overall efficiency, the water from a radiator network can be heated up in a pre-heater, where combustion gases cool down and release heat to the circulating water entering the water tank. The condenser can be located inside the boiler, in which case the condenser should be constructed with a finned type shell and a tube heat exchanger. If located outside the boiler, the condenser can utilize corrugated plate heat exchangers, which are extremely compact and effective. The temperature difference between condensing vapor and cooling water is larger when utilizing an outsider condenser.

3. Modeling integrated RSE system for performance assessment

Simplified models to predict the performance of fuel cells (SOFC and PEM), internal combustion engines and Stirling Engines in the whole-building simulation programs have been developed and

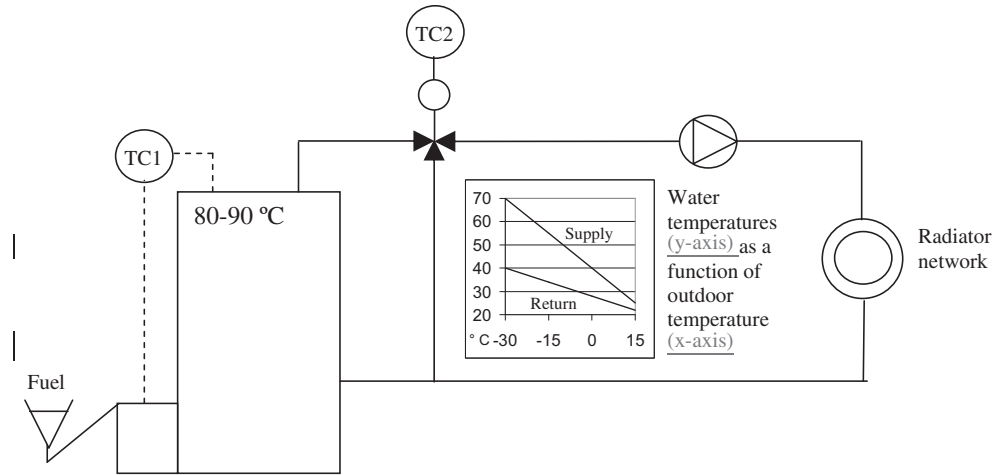


Fig. 2. Schematic diagram of a hydronic heating system [24].

implemented in the context of the IEA/ECBCS Annex 42 joint research program. Without attempting to characterize the thermodynamic behavior, the models account for the electrical and thermal performance of the system by way of parametric expressions, where steady-state electrical and thermal efficiencies are typically given as polynomials of electrical output, mass flow and temperature of the coolant water [25,26].

Other approaches have been developed by Onovwiona et al. [27], who present a parametric model for ICE micro-cogeneration, and Thiers et al. [28], who determine the thermal and electrical behavior of a wooden pellet SE micro-CHP system by way of experimentally verified piecewise linear equations. Moreover, Tzscheuschler et al. [29] have worked out a relation between the efficiencies of ICE and SE-based micro-cogeneration.

We introduce a thermodynamic model for an integrated RSE boiler, seeking similarity in principle with the IEA/ECBCS Annex 42 micro-cogeneration models [25]. Although simplified, this approach can estimate performance with sufficient accuracy in lieu of experimental data.

The Annex 42 device model for ICE/SE encompasses three control volumes: i) the energy conversion control volume, which expresses the conversion of fuel to electricity and heat in the engine under steady-state conditions; ii) the thermal mass control volume, which represents the thermal mass or the thermal capacitance of the engine as a whole, and iii) the cooling water control volume, which characterizes the heat exchange between the exhaust gas and the cooling water and all engine parts in immediate thermal contact. The reader is referred to Beausoleil-Morrison [23] and Ferguson et al. [30] for a detailed treatment.

The present application is principally a modification of the Annex 42 model for ICE/SE. Here, we have only two-control-volumes: one representing the energy conversion process as a whole, and the other the water domain inside the boiler. The water control volume makes it possible to deliver heat at various temperature levels, separately to hot water production and radiator network. The thermal loss via exhaust gases is one of the output variables of this model. The conversion control volume is not necessary if the changes in circulation water temperature are

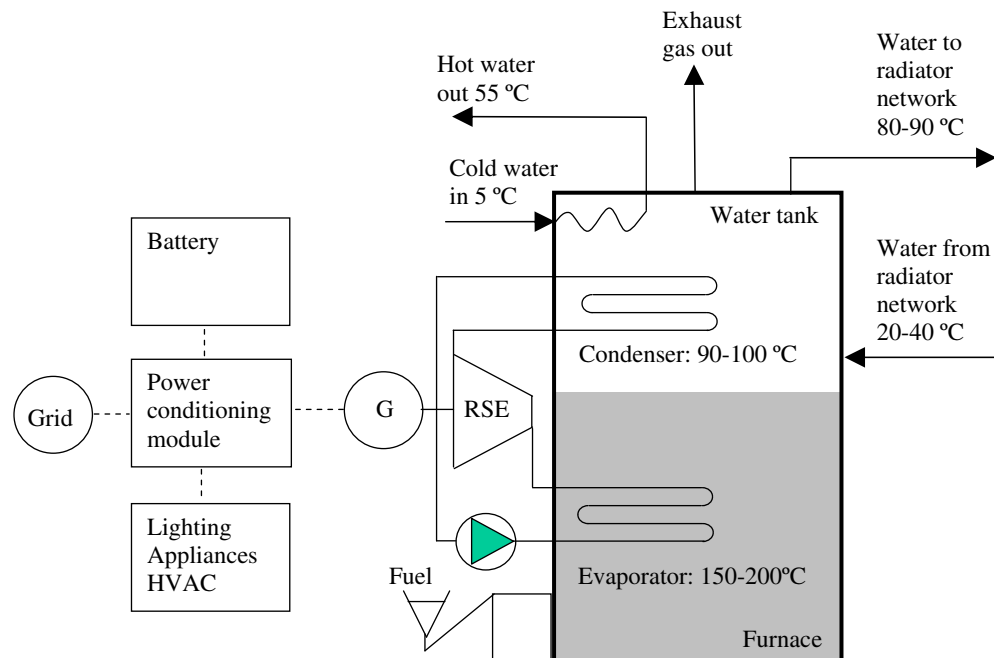


Fig. 3. Schematic diagram of the integrated RSE boiler system.

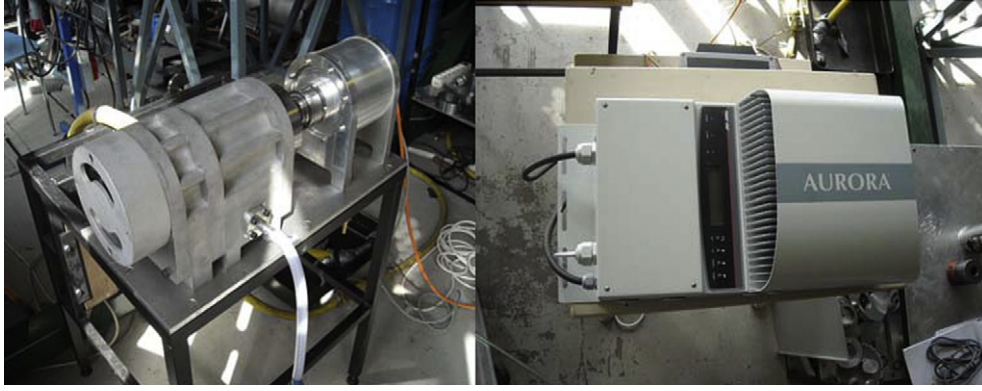


Fig. 4. Rotary steam engine (left) and power conditioning module (right).

moderately slow. It is, however, needed when the cold start procedure of the RSE plant is modeled. The control volumes of the model are depicted in Fig. 5.

The efficiencies of the integrated system are:

$$\text{Electrical efficiency } \eta_e = \frac{P_{\text{net}}}{\Phi_{\text{fuel}}} \quad (1)$$

$$\text{Thermal efficiency } \eta_{\text{th}} = \frac{\Phi_{\text{cond}}}{\Phi_{\text{fuel}}} \quad (2)$$

$$\text{Overall efficiency } \eta_{\text{tot}} = \frac{P_{\text{net}} + \Phi_{\text{cond}}}{\Phi_{\text{fuel}}} \quad (3)$$

In the above definitions, Φ_{fuel} is the thermal flow from fuel calculated from lower heating value (LHV), Φ_{cond} is thermal flow from condenser to the water control volume, Φ_{exh} is the thermal loss through exhaust gases, $\Phi_{\text{skinloss,c}}$ is the skin loss, and P_{net} is the net electrical power.

Fig. 6 represents the equipment in the conversion control volume.

The overall energy balance of the conversion control volume is

$$\Phi_{\text{fuel}} - \Phi_{\text{cond}} - \Phi_{\text{exh}} - \Phi_{\text{skinloss,c}} - P_{\text{net}} = C_c \frac{dT_c}{dt} \quad (4)$$

where C_c is the heat capacity of the entire boiler system (excluding water) and T_c is the supposed average temperature at which the heat

transfer from the conversion control volume to the water control volume (Φ_{cond}) and to the surroundings ($\Phi_{\text{skinloss,c}}$) takes place. In practice, T_c can be assumed the same as the temperature of the exhaust gases after the evaporator, which makes it easy to measure when the model is calibrated experimentally. The water volume and thus the heat capacity of the spiral tube evaporator are relatively small. For a 25 kW boiler the heat capacity of water plus a metallic tube is approximately 50 kJ/K, which means that the cold start of the evaporator from the initial temperature of 20 °C to the operation temperature 160 °C takes about 5 min. Constant temperature is maintained in the boiler (to produce hot water) most of time, however, as a result of which the conversion control volume can be assumed stationary (no heat capacity) in most practical applications.

Evaporator temperature T_{ev} is a design parameter for an RSE, limited by the design pressure of the system. The steam outlet from the evaporator is at saturated state (1), where $T_1 = T_{\text{ev}}$. The expansion is characterized by way of expansion ratio, which is another technology-specific design parameter. Expansion ratio is defined as $\gamma = v_2/v_1$, where v_1 and v_2 are the specific volumes for the vapor before and after the expansion. Here, the specific volume v_1 represents saturated state, as well.

The expansion volume ratio of an RSE is a fixed design value and is optimal only for a certain boiler and condenser conditions where the specific volume v_1 corresponds to the specific volume at boiler conditions and specific volume v_2 corresponds to the condenser conditions.

The available work of one cycle of an RSE can be calculated based on the mass of vapor m which is filling the RSE. Sweep volume $V_1 = mv_1$ is filled with saturated vapor when the environment (vapor from boiler) expands at constant pressure p_1 and does work which corresponds to

$$W_1 = \int_0^1 -pdV = -p_1 V_1 \quad (5)$$

The inlet valve is then closed and the vapor expands nearly isentropically from pressure p_1 to p_2 . Taking into account that for isentropic change $dH = Vdp$ and $d(pV) = pdV + Vdp$ this expansion work can be calculated as

$$\begin{aligned} W_{1,2} &= - \int_1^2 pdV = - \int_1^2 d(pV) + \int_1^2 Vdp = -p_2 V_2 + p_1 V_1 \\ &+ H_2 - H_1 = m(p_1 v_1 - p_2 v_2 + h_2 - h_1) \end{aligned} \quad (6)$$

where m is mass of vapor within the RSE.

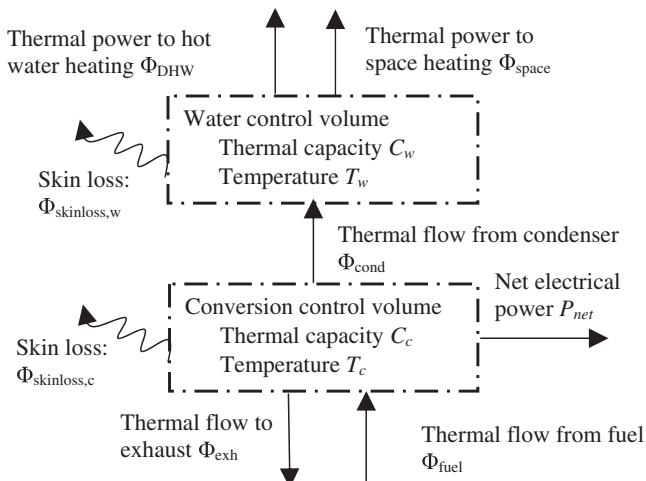


Fig. 5. Control volumes of the integrated RSE boiler system model.

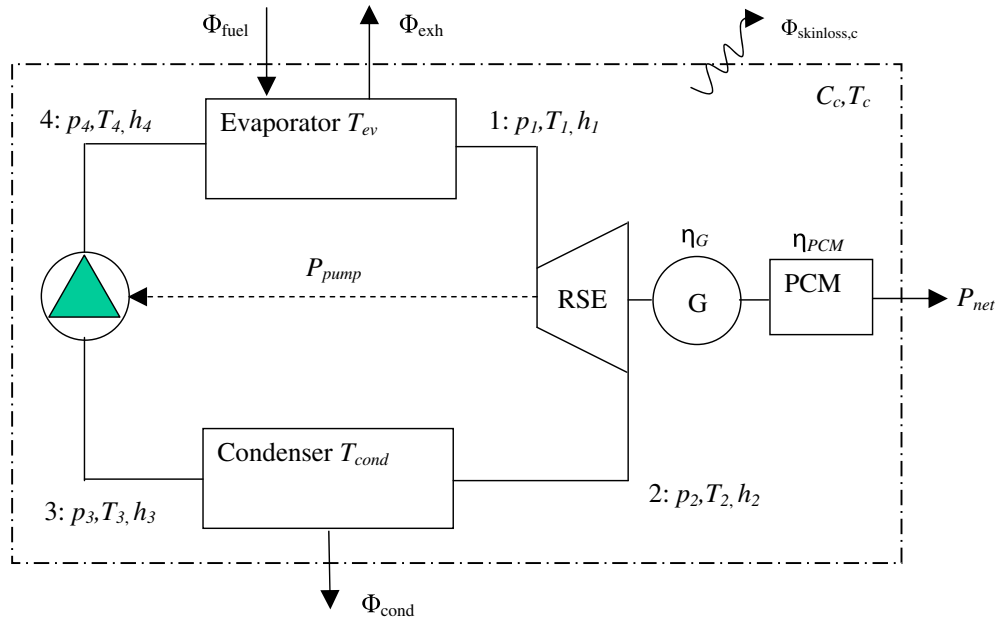


Fig. 6. Conversion control volume of the integrated RSE boiler system.

After the expansion the outlet valve is opened, and the vapor flows and expands to the condenser pressure p_3 . This expansion work is wasted, because it does not do any useful work to the RSE. Finally the RSE removes rest of the vapor (volume V_2) to the condenser pressure p_3 and does work to the environment:

$$W_3 = - \int_3^0 p dV = p_3 V_2 \quad (7)$$

The total work done to the RSE per one working cycle is $W_{\text{tot}} = -(W_1 + W_{1,2} + W_3)$. When the rotational speed of the RSE is n , the average mass flow is $q_m = nm = n\rho_1 V_1$ and the total power $P_{\text{tot}} = nW_{\text{tot}}$.

The isentropic expansion of vapor from the boiler pressure p_1 to the final condenser pressure p_3 corresponds to the enthalpy difference Δh_{is} . The efficiency of the RSE process compared to the isentropic expansion process can be expressed as

$$\eta_{\text{is}} = \frac{nW_{\text{tot}}}{q_m \Delta h_{\text{is}}} = \frac{q_m \frac{W_{\text{tot}}}{m}}{q_m \Delta h_{\text{is}}} = \frac{W_{\text{tot}}/m}{\Delta h_{\text{is}}} \quad (8)$$

The available power can then be calculated based on the isentropic expansion of vapor

$$P_{\text{tot}} = \eta_{\text{is}} q_m \Delta h_{\text{is}} = q_m (h_1 - h_2) (= nW_{\text{tot}}) \quad (9)$$

The condensing phase takes place at the pressure after expansion p_3 (at maximum $p_3 = p_2$) and condensate temperature $T_{\text{cond}} = T_3$ (saturated), which is also a design parameter. Condensate temperature remains constant ($T_{\text{cond}} = T_4$), whereas the circulation pump increases the liquid pressure to $p_4 (= p_1)$. Corresponding enthalpies (h_3 and h_4) are functions of the above temperatures and pressures.

Thermal flow through condenser is defined as

$$\Phi_{\text{cond}} = q_m (h_2 - h_3) \quad (10)$$

The net electrical output of the system is calculated from

$$P_{\text{net}} = \eta_{\text{ec}} q_m (h_1 - h_2) - P_{\text{pump}} \quad (11)$$

where η_{ec} is the electricity conversion efficiency and P_{pump} is the electrical demand of the circulation pump. The power demand of the circulation pump is

$$P_{\text{pump}} = \frac{q_m (h_4 - h_3)}{\eta_{\text{pump}}} \quad (12)$$

where η_{pump} is the efficiency of the circulation pump. Electrical conversion efficiency is the product of the efficiencies of generator (η_G) and power conditioning module (η_{PCM}) $\eta_{\text{ec}} = \eta_G \eta_{\text{PCM}}$.

Thermal flow through evaporator is

$$\Phi_{\text{ev}} = q_m (h_1 - h_4) \quad (13)$$

Skin loss from the conversion control volume is calculated from

$$\Phi_{\text{skinloss,c}} = (UA)_c (T_c - T_{\text{amb}}) \quad (14)$$

where $(UA)_c$ is the overall heat transfer coefficient of the conversion control volume and T_{amb} is the ambient temperature.

Heat flow from the exhaust gases is calculated based on the specific enthalpies of a moist exhaust gas

$$\Phi_{\text{ex}} = q_a [h(T_b) - h(T_{\text{exh}})] \quad (15)$$

where T_b is the combustion temperature and q_a is the mass flow of exhaust gases. The combustion temperature depends on the air factor, which is nearly constant for standard combustors. The final temperature of the exhaust gases after the evaporator, T_{exh} , is dependent on the heat transfer area of the evaporator (and possibly a pre-heater). Expected minimum temperature difference (pinch temperature difference) between the water inlet into the boiler and the exhaust gases is set at approximately 50 °C, to prevent the heat transfer areas from becoming unreasonably large and to prevent the exhaust gases from condensing. This will fix the minimum final temperature of the exhaust gases.

An ON/OFF controlled boiler operates at nominal (design) power when the system is switched on. In that case, the fuel feeding mechanism is fully open or closed and the fuel input corresponds to the fuel capacity. The thermal flow transferred to water circulation in the condenser is referred to as nominal thermal power of the

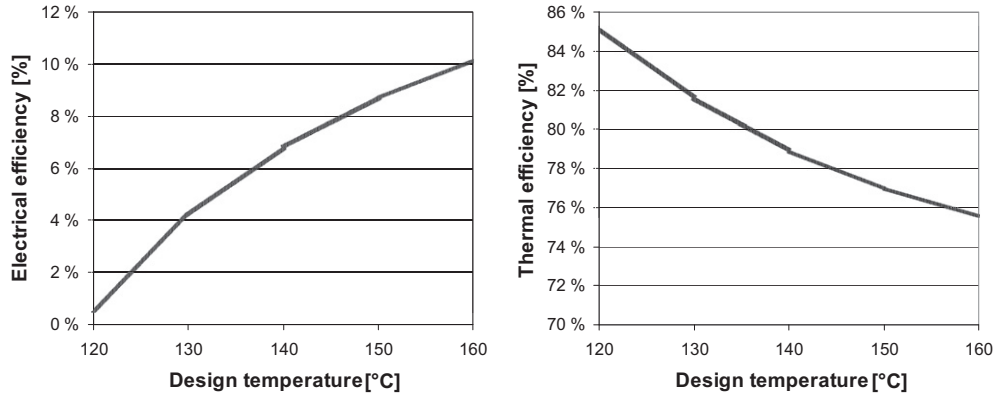


Fig. 7. Electrical and thermal efficiencies as a function of design (evaporator) temperature.

system. The net electrical output in the same conditions is the nominal electrical output of the system.

A part load operation is produced by throttling the pellet feed-in mechanism so that the fuel input represents a certain fraction of the fuel capacity. Part load ratio is defined as $r = \Phi_{\text{fuel}} / \Phi_{\text{fuel,nom}}$, where $\Phi_{\text{fuel,nom}}$ is the fuel capacity of the boiler. Here, the mass flow of the exhaust gas is assumed to be directly proportional to fuel capacity and the heat transfer coefficient of exhaust gases proportional to the Reynolds number as $\text{Re}^{0.8}$.

The water control volume is a fully mixed water tank, with separate heat exchangers for domestic hot water production and a radiator network. The overall energy balance is

$$\Phi_{\text{cond}} - \Phi_{\text{space}} - \Phi_{\text{DHW}} - \Phi_{\text{skinloss,w}} = C_w \frac{dT_w}{dt} \quad (16)$$

where Φ_{space} is the thermal flow from the water control volume to the radiator network, Φ_{DHW} is the thermal flow to the water heater, $\Phi_{\text{skinloss,w}}$ is the thermal flow to the surroundings, C_w is the heat capacity of the tank water and T_w is the water temperature.

The heat flow to the surroundings (skin loss) is

$$\Phi_{\text{skinloss,w}} = (UA)_w (T_w - T_{\text{amb}}) \quad (17)$$

where $(UA)_w$ is the overall heat transfer coefficient and T_{amb} is the ambient temperature.

4. Computational analysis

4.1. Numerical characterization of a 17 kW_{th} RSE boiler system

The conversion control volume model introduced in Section 3.3 is employed to characterize the performance of an RSE in an integrated boiler, the nominal thermal flow from the condenser to the water control volume being 17 kW_{th}. This is the thermal power of a common, pellet-fueled hydronic boiler (Ariterm Biomatic+). The estimated physical dimensions of an integrated pellet boiler are 935 × 608 × 1755 mm, when Ariterm Biomatic +20 is considered the reference model and the RSE is located on the boiler. The analysis is conducted in a spreadsheet application, where the nominal thermal output, evaporator and condenser temperatures, expansion ratio, nominal rotation speed, combustion temperature, temperature difference between exhaust gases and the water pre-heater, maximum allowable exhaust gas temperature after the boiler, and part load ratio can be given as input parameters. The final result of this analysis is the part load electrical, thermal and overall efficiency of the integrated boiler and its electrical and thermal output at a given part load factor.

The analysis is iterative by nature; the desired thermal flow in the condenser ($\Phi_{\text{cond}} = 17 \text{ kW}_{\text{th}}$), expansion ratio ($v_2/v_1 = 4$) and exhaust gas temperature ($T_{\text{exh}} = 150^\circ\text{C}$, which is fixed by the pinch temperature difference between the water pre-heater and exhaust gases (50°C) are obtained by looking for such final pressure of the expansion (p_2), fuel input (Φ_{fuel}), and equivalent sweep volume (V_s) that the pre-conditions become fulfilled. The steam tables implemented in the spreadsheet application as user-defined functions are exploited.

Quasi steady-state operation is assumed, that is, the operation at a nearly constant temperature is of interest and the thermal mass of the control volume is neglected. The temperature difference between the condenser and water is 5°C and that between exhaust gases and steam 50°C , the combustion temperature being 900°C . An isentropic efficiency of 90% is presumed. The rotational speed is the specific rotational speed of the engine (1500 rpm, 25 L/s), and the power demand of the circulation pump as well as the electricity conversion losses are neglected as marginal. The skin loss from the conversion control volume to the room is considered small, and all the losses are included in the exhaust losses.

The overall efficiency of the RSE micro-cogeneration system depends on the condenser temperature. The lower the condenser temperature, the higher the overall efficiency is. Since the boiler must be able to maintain the water tank temperature at 90°C , the minimum condenser temperature in the present application is 95°C , which results in the overall efficiency of approximately 87% (LHV). One of the advantages of an RSE is low design pressure (6 bar). According to Finnish law 869/1999 for pressure vessels, if the pressure is less than 10 bar, the system can be operated by a customer without periodic inspection of pressure devices. The inside diameter of the tubes in the present application is 20 mm and the length of the tube is 60 m, which corresponds to the volume of 19 L. Hence, the pressurized volume is not subjected, for example, to the North American National Board, Synopsis of Boiler and Pressure Vessel Laws, Rules and Regulations, either.¹ The design pressure of 6 bar corresponds to the evaporator temperature of 158°C . The electrical and thermal efficiencies as a function of evaporator temperature are illustrated in Fig. 7.

Fig. 7 suggests that increasing the design temperature boosts the electrical output, although at the expense of thermal output. This is a direct consequence of the increased evaporator pressure. The overall efficiency based on both thermal and electrical output remains approximately constant.

¹ Pressure vessels subject to rules and stamping: a) 5 cubic feet (141.6 L) in volume and 250 psig (17.2 bar) pressure; b) 1–1/2 cubic feet (42.5 L) in volume and 600 psig (41.4 bar) pressure; c) an inside diameter of 6 inches (15 cm) with no limitation on pressure.

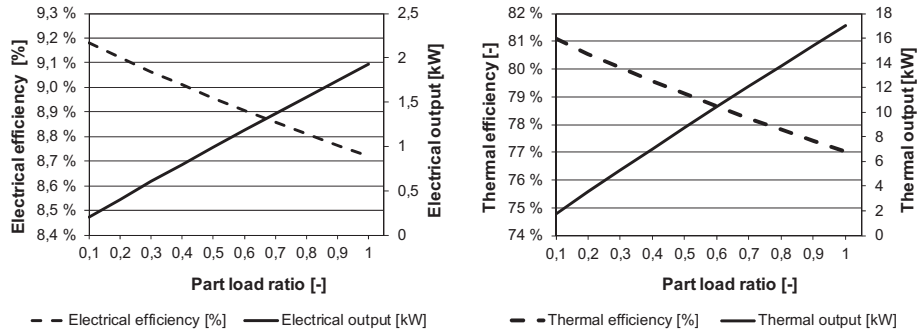


Fig. 8. Part load performance of the integrated RSE boiler system.

The present application is accommodated to evaporator temperature of 150 °C, which results in the design pressure of 4.76 bars, and the condenser temperature of 100 °C. This is a compromise between efficiency and practicality, which makes it possible to design a safe and inexpensive micro-cogeneration plant. Here, the overall efficiency of 86% is obtained, which is of the same magnitude as the thermal efficiency of corresponding pellet boilers (of a Finnish manufacturer) without electricity generation. The part load performance is summarized in Fig. 8.

In Fig. 8, the correlation between part load ratio and efficiency is linear, because the mass flow of exhaust gas is directly proportional to the fuel input.

4.2. Performance of the integrated RSE boiler in a residential application

The IDA-ICE whole-building simulation tool, which has been validated for example in Travesi et al. [31] and Loutzenhiser et al. [32] are employed in combination with a spreadsheet-based post-processing application to determine the operational hours, the electric power shortage and surplus for a single-family house equipped with a 1.925 kW_e/17 kW_{th} integrated RSE boiler. The volume of the water tank is 140 L. The reference boiler is a standard pellet-fired boiler without an RSE, operating at the thermal efficiency of 86% (LHV) and nominal thermal output of 17 kW_{th}.

The target building is a one-floor single-family house with a heated area of 135.6 m² occupied by four persons between 5:00 P.M. and 8:00 A.M. The reference location is Helsinki, Finland, and the weather data collected at the Helsinki-Vantaa International Airport during 1980–2009 is used. The reader is referred to Jylhä et al. [33] for further details. The building is equipped with a central mechanical exhaust ventilation system (constant air flow of 49.5 L s⁻¹) with no heat recovery. The heating system is programmed to maintain the room temperature at 21.5 °C. The building does not contain a cooling system, allowing the temperature to rise during summer months. The heat transfer coefficients (*U*-values) for walls, windows, floors and ceilings are 0.42, 2.2, 0.3, and 0.24 W/m²K, respectively.

The electrical demand is the sum of the power requirements of appliances (including fans and pumps) and lighting. The hourly profile is estimated on the basis of everyday experience. The annual level of electrical consumption is acquired from Haulio [34]. The daily profile is shown in Fig. 9.

The demand of domestic hot water is evaluated, relying on the general estimate that four occupants consume 60 L per person per day [35]. The specific water flow of a tap and a shower is 0.2 L/s [36], and the estimate of the Finnish Energy Agency (Motiva) for the proportion of hot water in the total water consumption is 40%. The water is heated from 5 °C to 55 °C [35]. Assuming that showers are in use 5 min at 7:00 A.M., 8:00 P.M., 9:00 P.M. and 10:00 P.M. and

the taps 10 min at 6:00 A.M., 6:00 P.M. and 11:00 P.M., the total consumption of hot water is 240 L. Outside the heating period, thermal energy is still needed to produce hot water, and a high temperature must be maintained in the boiler. In stand-by conditions, heat transfer between the boiler and the surrounding air (skin losses) must be compensated.

The skin losses are estimated with an assumption that the temperature of the boiler envelope is constant 40 °C (313 K) and the surrounding temperature is 25 °C (298 K). The height, width and length of the boiler are 1.555 m, 0.608 m, and 0.935 m, respectively. The skin losses take place in terms of free convection from the vertical surfaces and the top of the boiler, the lower surface being insulated. The surface temperature is assumed to be uniform and the room air quiescent. The skin loss is calculated from

$$\dot{\Phi}_{\text{skinloss}} = \alpha A_s (T_s - T_{\text{amb}}) \quad (18)$$

where α is the convective heat transfer coefficient from the surface to the surroundings, A_s is the area of the boiler envelope and T_s is the surface temperature of the envelope. The heat transfer coefficient is determined on the basis of the Nusselt number, which, in turn, is a correlation of the Rayleigh (*Ra*) and Prandtl (*Pr*) numbers [37]. The thermal properties of air at 300 K are used. The estimated skin loss is now 283 W and the daily thermal demand 21 kW h (1 kW h = 3.6 MJ), where the proportion of skin losses is 7 kW h.

Two operational strategies are first investigated for the coldest day of the year (February 6), to demonstrate the control volume temperatures and electricity production patterns of the integrated boiler system. The peak electrical demand is 1.2 kW and the peak thermal demand 23.9 kW (space heating and domestic hot water). In the first control strategy, the ON/OFF operation, the boiler is switched on when the temperature of the water control volume drops under 80 °C and is switched off when this temperature

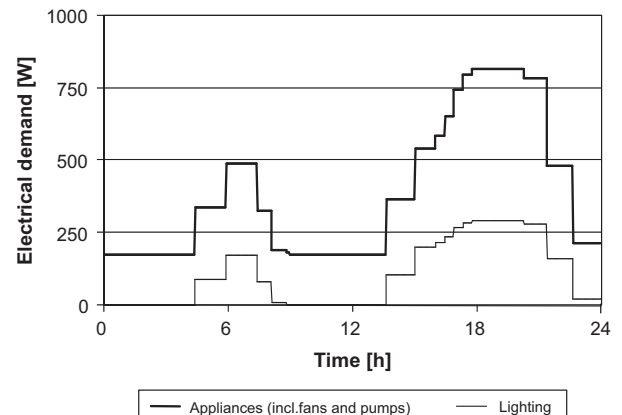


Fig. 9. Daily profile of electrical demand.

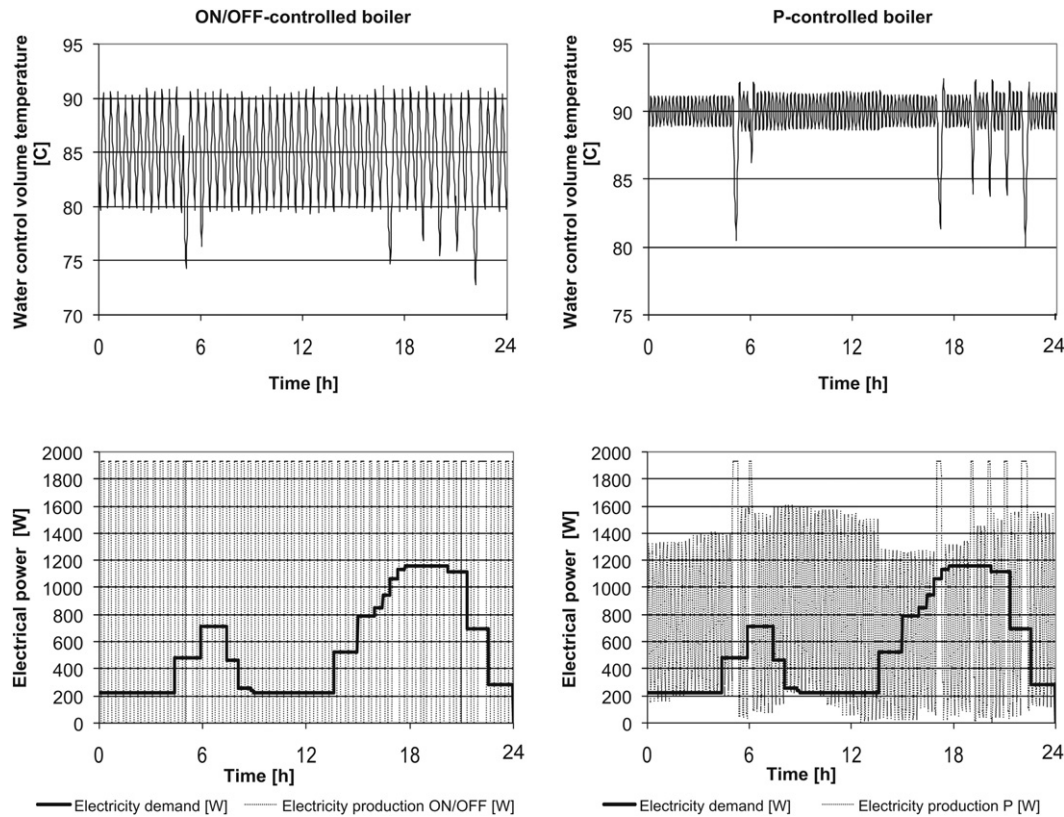


Fig. 10. Control volume temperatures and electricity production (the coldest day).

exceeds 90 °C. In the second strategy, a set point temperature of 90 °C is maintained in the water control volume and a simple proportional (“P”) control is applied, where the control parameter (part load ratio) is estimated on the basis of the previous time step. The water control volume temperatures and electrical outputs are depicted in Fig. 10.

As seen in Fig. 10, the water temperature reacts to the peak thermal demands caused by the increased hot water demand, and electricity production follows. At the given parameters, the water temperature remains over 70 °C, which is the design condition for supply water temperature. The design thermal output of 17 kW_{th} proves to be a little undersized for the present application. The data in Fig. 10 also implies that P-control never reaches the set point, but the temperature of the water control volume fluctuates around the desired temperature. When P-controlled, the integrated boiler operates throughout the day with on-site power covering 74% of total electricity demand, whereas with ON/OFF control this share is reduced to only 44%. The P-controlled system delivers 11 kW h into the grid, against 14 kW h for the ON/OFF controlled system. The thermal energy production and the consumption of fuel remain the same for both approaches.

The annual energy performance and operational hours are summarized in Table 1. The integrated boiler is ON/OFF controlled, and the performance of a Stirling Engine (SE) is shown for comparison. The SE is assumed to be integrated into a boiler identical to that of the RSE. The condenser and evaporator are now substituted by heat exchangers, helium acting as the working gas. The nominal output of the integrated SE boiler is 1.64 kW_e/15.9 kW_{th}, the electrical, thermal and overall efficiencies being 7.3%, 71%, and 78.3%, respectively. The performance of the SE boiler remains inferior to that of the corresponding RSE boiler, because i) the design of the heat exchanger is limited by the

physical size of the boiler and ii) heat transfer is less effective for the SE boiler. The latter stems from that the convective heat transfer coefficient between the tube and helium flow is smaller than in evaporating and condensing water in the RSE system. When using similar heat exchangers, less heat is now transferred into the cycle process, and the final exit temperature of combustion gases remains higher.² On the other hand, helium remains at a higher temperature than the condensing temperature of water, because a sufficient temperature has to be maintained in the water control volume.

To evaluate the economic viability of the integrated RSE boiler, the break-even incremental costs related to the prime mover (RSE) and heat exchangers were calculated at different payback periods (5–10 y) at optimistic, realistic and pessimistic scenarios. Using the prices provided by the Finnish Energy Agency (Motiva) for electricity and wooden pellets,³ 12.5 c/kWh and 4.8 c/kWh, respectively, the maximum allowable incremental costs are depicted in Fig. 11.

Fig. 11 indicates that the integrated boiler creates savings over time, but the economic viability is sensitive to both discount (*r*) and buyback rates (BRate). Buyback rate depends on country-specific feed-in tariff. In 2010, there was no feed-in tariff in Finland, whereas in Germany 11.67 c/kWh was paid for the first 150 kW of output generated by biomass. This electricity was produced by biomass installations with the capacity of less than 150 kW [38].

² Helium mass flow 0.1 kg/s, tubular heat exchanger, outside diameter 12 mm, inside diameter 10 mm. Cold side (water): 100 tubes, length 250 mm; Hot side (furnace): 3 tubes, length 2620 mm.

³ The moisture content of wooden pellets is typically 6–12% and the ash content less than 1% (e.g. www.cemagro.fi/pelletti.pdf).

Table 1
Annual energy performance of alternative boilers.

	Standard boiler	Integrated boilers	
		RSE	SE
Operating hours [h/y]	1,274	1,274	1,366
Generated thermal energy [kWh/y]	21,666	21,666	21,666
On-site electricity [kWh/y]	0	2,453	2,241
Electricity from the grid [kWh/y]	4,711	3,246	3,146
Electricity to the grid [kWh/y]	0	989	679
Fuel [kWh/y]	25,268	28,129	30,532
Percentage of on-site electricity [%]	0	31	33

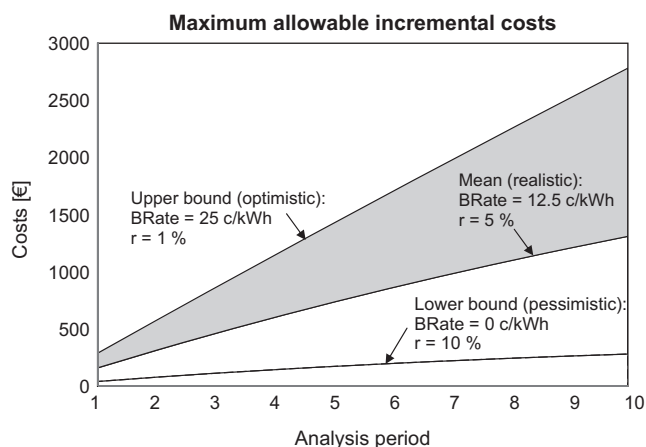


Fig. 11. Maximum allowable incremental costs of the integrated RSE boiler.

Discount rate (r) varies between 1 and 10%. When assessing residential heating investments, the expected payback time is commonly less than five years. Hence, an incremental price of no more than € 1500 (€ 780/kW_e) above standard boilers appears to be economically justified even in the optimistic scenario. This is significantly less than, for example, the specific investment cost of an optimized ORC micro-cogeneration system presented by Quoilin et al. [21]. The above result is applicable to the present analysis only, but is nonetheless an indicative guideline for the further development of the integrated RSE system.

5. Conclusions

In this paper, a pellet-fired boiler is proposed to be integrated with a rotary steam engine to construct a micro-cogeneration system. Electrical efficiency of 9%, thermal efficiency of 77% and an overall cogeneration efficiency of 86% (all in terms of LHV) can be obtained using the design temperature of 150 °C, which allows the design pressure to be kept acceptable in terms of safety and cost-effectiveness. To improve the electrical efficiency, the evaporator temperature is allowed to go up to 300 °C. In a performance assessment study for a single-family house located in Finland, the integrated boiler generated 2453 kW h of electricity during 1274 h/y operating hours of which 1465 kW h/y was consumed (31% of 4711 kW h/y) and 989 kW h/y was exported. The economic viability analysis suggests that the incremental price of integrated boilers no more than € 1500 above standard boilers is acceptable. In the future, a prototype will be constructed and both laboratory and field tests conducted, to find out its performance in realistic conditions and to gather operational experience for the further development of the prototype to a commercial product.

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